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### EVALUATING MANUFACTURING AND ASSEMBLY ERRORS IN ROTATING MACHINERY TO ENHANCE COMPONENT PERFORMANCE

**Irem Y. Tumer \***

Computational Sciences, CSC  
Systems Health and Safety  
NASA Ames Research Center  
Moffett Field, California 94035  
Email: itumer@mail.arc.nasa.gov

**Edward M. Huff**

Computational Sciences  
Systems Health and Safety  
NASA Ames Research Center  
Moffett Field, California 94035

#### ABSTRACT

Manufacturing and assembly phases play a crucial role in providing products that meet the strict functional specifications associated with rotating machinery components. The errors resulting during the production of such components are correlated with the vibration and noise emanating from the final system during its operational lifetime. Vibration and noise are especially unacceptable elements in high-risk systems such as helicopters, resulting in premature component degradation and an unsafe flying environment. In such applications, individual components often are subject to 100% inspection following production, as well as during operation through rigorous maintenance, resulting in increased product development cycles and high production and operation costs. In this work, we focus on providing designers and manufacturing engineers with a technique to evaluate vibration modes and levels for each component or subsystem prior to putting them into operation. This paper presents a preliminary investigation of the correlation between manufacturing and assembly errors and vibrations, using an experimental test rig. A factorial design is used to study the effects of: 1) different manufacturing instances; 2) different assembly instances; and, 3) varying shaft speeds. The results indicate a correlation between manufacturing or assembly errors and vibrations measured from accelerometers. Challenges in developing a tool for design and manufacturing engineers are identified, followed by a discussion of future work.

#### INTRODUCTION

The intended function of a component can be compromised if there are errors during production, which result in undesired side effects. Two of the significant factors that cause undesired vibrations are manufacturing and assembly errors. In this work, knowledge of the correlation between these errors and vibrations is seen as a crucial piece of information. To this end, we explore the relationship between manufacturing and assembly variations, and vibration patterns in rotating machinery.

The overall goal in this work is to develop practical tools to fit NASA's overall goal of designing and developing safer aircraft with shorter turn-around times, reduced cost, and increased quality. A typical production cycle for rotating machinery components involves regular quality control steps to assure that the specifications are met satisfactorily. In particular, rotating components in high-risk applications such as rotorcraft transmissions are subject to an intense inspection process during manufacturing, before and after assembly, as well as during operation (maintenance). In such cases, 100% inspection of the parts' manufacturing and assembly tolerances are typically required, increasing the development time and the cost of producing such parts. A prediction of potential deviations from the intended functional requirements will not only reduce safety risks by avoiding premature failure, but also shorten the product cycle by avoiding scrap, rework, and inspection, as well as decrease costs associated with unplanned maintenance, hence reducing the overall cost of producing and operating such products (Tumer et al., 1998; Tumer et al., 2000a; Tumer et al., 2000b).

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\* Address all correspondence to this author.

## Tools for Design and Manufacturing

Designers and manufacturing engineers use a combination of tools (six-sigma, inspection, statistical process control, Taguchi's robust design method, error budgeting, etc.) to assess and eliminate variation, with the goal of producing higher quality parts with less scrap or rework, hence, reducing the time and cost of product development (Bothe, 1997; Bralla, 1999; Carter, 1997; Chen and Thornton, 1999; Frey and Otto, 1997; Tata and Thornton, 1999). However, there is still a continuing need to develop products with better performance in faster and less costly ways. To address this problem, many researchers in design and manufacturing have been working towards developing tools to help designers and manufacturing engineers. The idea of predicting and controlling variation has resulted in several promising methods (Frey and Otto, 1997; Chen and Thornton, 1999; Kazmer and Barkan, 1996; Suri and Otto, 1999). Design tools that contain manufacturing variation information have been shown to facilitate the design process by reducing the number of iterations involved, estimating critical parameters, and identifying possible problems in terms of quality and cost (Bralla, 1999).

In this work, the nature of the information passed onto the design and manufacturing engineers differs from standard variation information. Rather, this work is based on the idea that the manufacturing process leaves a fingerprint on manufactured product surfaces, which can be investigated experimentally to determine the operating importance of such physical sources of variation. The underlying notion in this work is that such variations only become crucial when the performance of the system in which it is installed is directly affected by the nature and severity of these variations. Previous work focused on developing tools to extract this information in an accurate fashion, in order to help close the information gap between design and manufacturing (Tumer et al., 2000a; Tumer et al., 2000b). This paper extends the idea to the aerospace framework (Kromholtz and Condra, 1993; Soni and Narang, 1997), where the safe and cost-effective performance of rotating components is dependent on the quality of the products resulting from the manufacturing and assembly phases.

## Characteristics of Ball Bearing Defects

In this paper, ball bearings are selected as the initial focus of study because of their tendency to contain waviness and roughness errors on their inner and outer raceway surfaces, as well as their tendency to move within the bearing housing due to assembly errors. The detection of defects in ball bearings is an important part of the work discussed in this paper.

A defect-free ball bearing has perfectly circular inner and outer races, constant thrust loading, and no radial loads (Meyer, 1980; Shigley and Mischke, 1989). Any deviation from these intended functions is a defect. The most typical failures include point defects on the outer race, inner race, the rolling elements,

and the cage: empirical formulas exist to compute the frequency at which these defects would appear in a vibration spectrum, usually a function of the shaft speed, ball diameter, number of balls, pitch diameter, and the contact angle (Braun, 1980; Mitchell, 1993; Taylor, 1980; Wowk, 1991). The relative amplitudes of these vibrational frequencies indicate the change in the bearing condition.

As opposed to point defects, problems due to manufacturing and assembly errors are difficult to isolate as single frequencies. Assembly errors are typically due to the bearing being mishandled after manufacturing, resulting in an improper mount or fit in the bearing housing for a system, or an improper mount or loading within the bearing cage. Such errors can result in looseness, unbalance, or misalignment, which are phenomena that typically result in high first, second, and third harmonics (Brandlein and Eschmann, 1999; Wowk, 1991). Manufacturing errors, defined here as surface irregularities on bearing surfaces (typically under a micrometer in magnitude, not including form errors), are present in the form surface roughness and surface waviness (Braun and Datner, 1979; Harris, 1991; Ono and Okada, 1998; Su, 1993; Wensing and Nijen, 1996; Whitehouse, 1994). Surface roughness patterns are high-frequency (noise-like) components, resulting in high-frequency components in the vibration spectrum. Waviness patterns are low-frequency components with a periodic pattern, resulting in low-frequency components in the vibration spectrum.

Controlling component waviness and other types of errors from manufacturing, distortion, or damage occurring while the bearing is assembled to a machine, is a high priority, since the effects of such errors on machine vibration and noise can be significant. Vibration testing and waviness testing are typically performed on components as part of quality control, but typically, overall levels and thresholds are used for determining the quality of such components (Baldanzini and Beraldo, 1999; Chong and Yi, 1999; Harris, 1991). However, these testing methods depend heavily on the signal processing algorithms used in decomposing the vibrations signals, which often result in inadequate information. One of the standard methods and techniques is used in this paper for a preliminary look at the data.

## EXPERIMENTAL SETUP AND PREPARATION Mechanical Apparatus

The experimental test rig used in this experiment, the Machinery Fault Simulator (MFS), is shown in Figure 1. The product is currently marketed by SpectraQuest as a teaching aid, and was easily adapted for research use. As shown, the MFS is a desktop test rig, which houses a rotating shaft with weights to introduce balance faults, bearings attached at several locations to introduce bearing defects, and a gearbox attached to the main shaft by means of a belt to introduce gear defects. In the present study, most of the attachments provided with the device were



Figure 1. Mechanical Fault Simulator (MFS) manufactured by SpectraQuest.

taken off or disabled, such as the shaft unbalancing discs, belt drive to an external transmission assembly, and so forth, leaving the shaft free to rotate without load. This first set of experiments involves the motor main shaft, and the two ball bearing assemblies supporting the main shaft. Several types of bearings are provided for testing by the manufacturer, including a set of healthy and faulty ball bearings. In this study, three normal ball bearings were used for three production runs, making a total of nine different bearing/assembly combinations in the experiment.

### Data Collection Apparatus

In order to maintain consistency with vibration data collected on NASA test rigs and research aircraft, two single axis Endevco accelerometers were mounted near the bearing housings. During each experimental run, accelerometer and temperature sensors were sampled at  $10\text{kHz}$  for  $2.5\text{sec}$ . using a data-acquisition system containing many of the same components used aboard the NASA test-rigs and research aircraft. The anti-aliasing filter was set at  $5\text{kHz}$ . The data collection process was controlled by a LabVIEW software package (ALBERT) specifically developed by NASA for test-rig operations (Huff et al., 2000b; Huff et al., 2000a).

### Experimental Design

An experiment is designed using a factorial layout in order to test the effect of manufacturing and assembly errors, as well as shaft speed, and their interactions, on the vibration signature collected from the bearings. Since the primary focus of the experiment was to examine the combined effects of manufacturing variations, assembly inaccuracies, and operating speed, a three-way factorial design was used which allows the assessment of interactions between the factors as well as their main effects using

analysis-of-variance (ANOVA) techniques. Unusual aspects of this form of research, however, are that: (1) the dependent variables are time series rather than univariate measurements, and (2) the time series feature of greatest interest is total power, which, in the case of acceleration, represents the variance of the series. In other words, unlike typical empirical studies using ANOVA, this analysis is inherently concerned with a comparison of variance metrics rather than central tendencies. In order to meet this challenge, an extrapolation was made to compare equality of treatment variances by use of the natural logarithm of the data (Bendat and Piersol, 1986; Montgomery, 1991).

In the present design, Factor A represents three fixed levels of shaft speed ( $30\text{Hz}$ ,  $45\text{Hz}$ , and  $80\text{Hz}$ ). Factor B represents three assemblies of the bearings and is considered a random factor since we are interested primarily in generalizing to the population of assembly operations. With regard to Factor B, the three re-assemblies are conceptualized as simulating typical parts replacement procedures where the slightest misalignment, misfit, or looseness of the bearings in their housing can cause undesired vibrations that can be hazardous to the entire system's health. A rigorous re-assembly protocol was imposed to emulate the strict procedures followed when replacing helicopter transmission gear units (Huff et al., 2000b). Factor C represents the different bearing stocks from which three individual bearings were selected. Bearings from different stocks are assumed to contain manufacturing variations on their surfaces. This factor is also treated as a random factor since the population from which the bearings were drawn is of interest, not the particular three bearings themselves. The three factors with three levels each result in 27 experimental test conditions. Each test condition is replicated 11 times to provide within cell variance for proper statistical analysis (Montgomery, 1991).

### Experimental Effects

The experiments described above are conducted so that, for each bearing component, all three speeds and three assembly instances are tested. So, component 1, for example, is the first bearing assembled into the bearing housing once, and used at speeds 1, 2, and 3, and then disassembled, and reassembled into the bearing housing a second time, and run at speeds 1, 2, and 3, and finally disassembled and reassembled a third time. Then component 1 is replaced with the second bearing component, and then replaced again with the third bearing component, and the same steps are followed to cover all 27 combinations of speeds, components, and assembly instances. As a result, for each component, the results are expected to be similar at the three speeds. The different bearings represent a random sample of manufacturing errors (waviness and/or roughness errors) on the outer race, inner race, or the ball surfaces. Similarly, point defects can be present either on the outer race, inner race, or ball surfaces. Power spectra should provide a picture of the frequency content

Table 1. COMPUTED DEFECT FREQUENCIES (all frequencies in Hz; harmonics at K=1,2,3,...).

$f_{shaft}$	$f_{cage}$	$f_{outer}$	$f_{ballspin}$	$f_{ballfault}$	$f_{IRcontact}$	$f_{inner}$
30	11.24	$K \times 89.93$	56.11	$K \times 112.2 \pm 11.24$	18.76	$K \times 150.0 \pm 30$
45	16.86	$K \times 134.9$	84.17	$K \times 168.3 \pm 16.86$	28.14	$K \times 225.1 \pm 45$
80	29.97	$K \times 239.8$	149.63	$K \times 299.3 \pm 29.98$	50.02	$K \times 400.2 \pm 80$

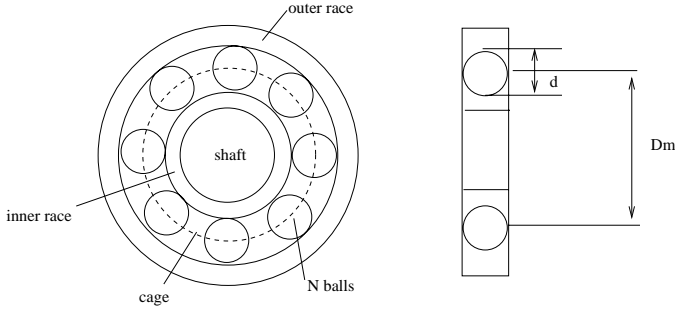


Figure 2. Ball Bearing Geometry.

of the vibration data, including the frequency components associated with the defects. One final note is on the assembly and component factors: because the assembly procedure is followed using predefined steps, the only effects likely to be detected are due to uncontrollable factors, which may or may not be repeated in the experiments. As a result, every time a new bearing component is used, disassembled and reassembled, the experiment might be introducing effects that are not repeatable, and it is unclear what factors will have led to differences, if any are detected.

### Computation of Defect Frequencies

The bearings used for this study are SKF bearings with  $N = 8$  balls, with a pitch diameter of  $D_m = 1.1228in$  (0.0285m), a ball diameter of  $d = 0.2813in$  (0.0072m), an inner ring thickness of  $t = 0.1083in$  (0.0210m), an inner race circumference of  $IR = 2.6437in$  (0.0671m), and an outer race circumference of  $OR = 5.0916in$  (0.1293m). The defect frequencies can be computed as a function of the shaft rotational speed, which is at a frequency of 30Hz, 45Hz, or 80Hz, depending on the run condition, and a function of the ball diameter, pitch diameter, and number of balls. A simple geometry of a ball bearing is shown in Figure 2 for reference.

A point defect on the outer race causes an impact each time a ball crosses it, resulting in  $N$  impacts for every revolution of the cage. Since the impacts are of short duration, the spectra will exhibit many harmonics. The computed cage frequency  $f_{cage}$ , and the corresponding outer race fault frequency  $f_{outerfault}$  are presented in Table 1 for each shaft speed condition. A flaw on a single ball will alternately strike the inner and outer races, resulting

in periodic forces at twice the ball spin frequency. Each impact is brief, and hence results in many harmonics. In addition, the contact points rotate at the cage precession frequency, which shows up as sidebands at  $\pm f_{cage}$ . The computed ball spin frequency  $f_{ballspin}$  and the corresponding ball fault frequency  $f_{ballfault}$ , are also presented in Table 1. Finally, a flaw on the inner race will be impacted by each of the  $N$  balls in sequence, at an inner race contact frequency  $f_{IRcontact}$ , each will be brief and lead to harmonics. In addition, the fault location rotates with the shaft, resulting in sidebands at  $\pm f_{shaft}$ . The resulting inner race fault frequency and the inner race contact frequency are presented in Table 1 for each speed.

The rotation of a ball with about its own axis is  $f_R$ , and the rotation of the inner race is  $f_{IR}$ . The vibration produced by waviness on the surface of a ball is computed as  $f_R$  times the number of waves per ball circumference. The vibration produced by inner race waviness is computed as  $f_{IR}$  times the number of waves per inner race circumference. Similarly, the vibration produced by outer race waviness is computed as the cage frequency  $f_{cage}$  times the number of waves per outer race circumference. The assumption is that any ball rolls over all the waves in the outer raceway in one cage revolution, resulting in a ball passage frequency over an individual wave cycle on the outer race. Unfortunately, there is no way of knowing the total number of waves on the outer race, inner race, or the ball surfaces, unless detailed measurements of the surface profiles are collected from the dismantled bearings. As a result, until such a measurement is made, there are no reported values for these expected frequencies. The measurement of the bearing surfaces will be performed at a future date, once the types of experiments to be conducted using these bearings have been exhausted.

## ANALYSIS OF RESULTS

### Power Spectral Analysis of Vibration Data

Before conducting a statistical analysis of the results, the vibration data are analyzed using standard signal processing techniques. The most standard means of analyzing vibration data is based on the power spectral analysis, which transforms the data into the frequency domain, isolating the frequency components of interest. The purpose is to investigate the frequency characteristics of the vibration data, and to isolate specific defect fre-

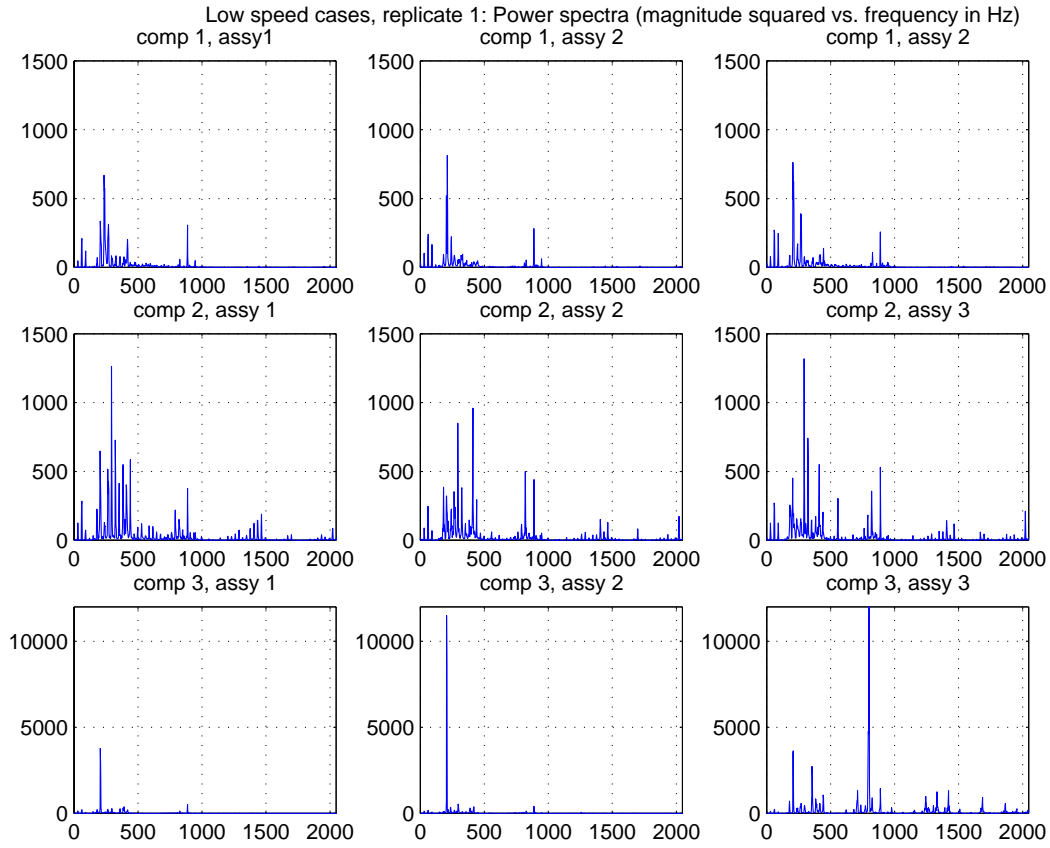


Figure 3. Power Spectral Analysis, Low-Speed Case, Replicate 1.

quencies. Despite many problems faced by spectral analysis, especially in the presence of time-varying trends, to be compatible with industry standards, this paper uses the standard technique (Tumer et al., 2000a). More advanced methods and metrics will be explored in a future study.

Figure 3 provides an overall picture of the frequency content for components 1, 2, and 3, assemblies 1, 2, and 3, for the low speed runs (30Hz). The top three spectra correspond to the first bearing component, assembled into the same housing three times. The next three plots correspond to the second bearing component, and the last three correspond to the third bearing component. Similar plots are studied for each speed, and each replicate. Figure 4 presents a comparison of the frequency content for component 1, assembly 1, for each of the three speeds.

An analysis of the power spectra indicates that the overall magnitudes of the frequency components are higher for component 2 than for component 1. This effect is repeated over all the replicates, and speeds. In addition, the spectra for component 2 indicate the presence of higher frequency components, whereas the spectra for component 1 are less significant in the higher frequencies. This pattern is indicative of a defect in component 2,

possibly in the form of surface waviness errors either on the outer race or the inner race. This conclusion will not be finalized until the inner and outer race surface profiles are measured and compared for the two bearings. Component 3, on the other hand, implies a different set of conclusions. The magnitudes of a couple of the harmonics are much higher in comparison to the spectra for the first 2 components. In addition, a lot more variability is present between the different replicates, possibly indicating an unbalance error within the bearing, since the frequency content is not repetitive or consistent.

Overall, the spectra from all the cases display the typical frequency content for rotating machinery, composed of the shaft rotational frequency and its many harmonics. Figure 4 shows a comparison of the spectra for each of the three shaft speeds. Notice that the amplitudes of the frequency components increase as the shaft rotational speed increases. The first, second, and third plots correspond to shaft speeds of 30Hz, 45Hz, and 80Hz, respectively, and show their corresponding harmonics. The presence of harmonics can also be indicative of potential misalignment or mechanical looseness problems, which can both be introduced due to assembly errors. In general, the magnitudes of

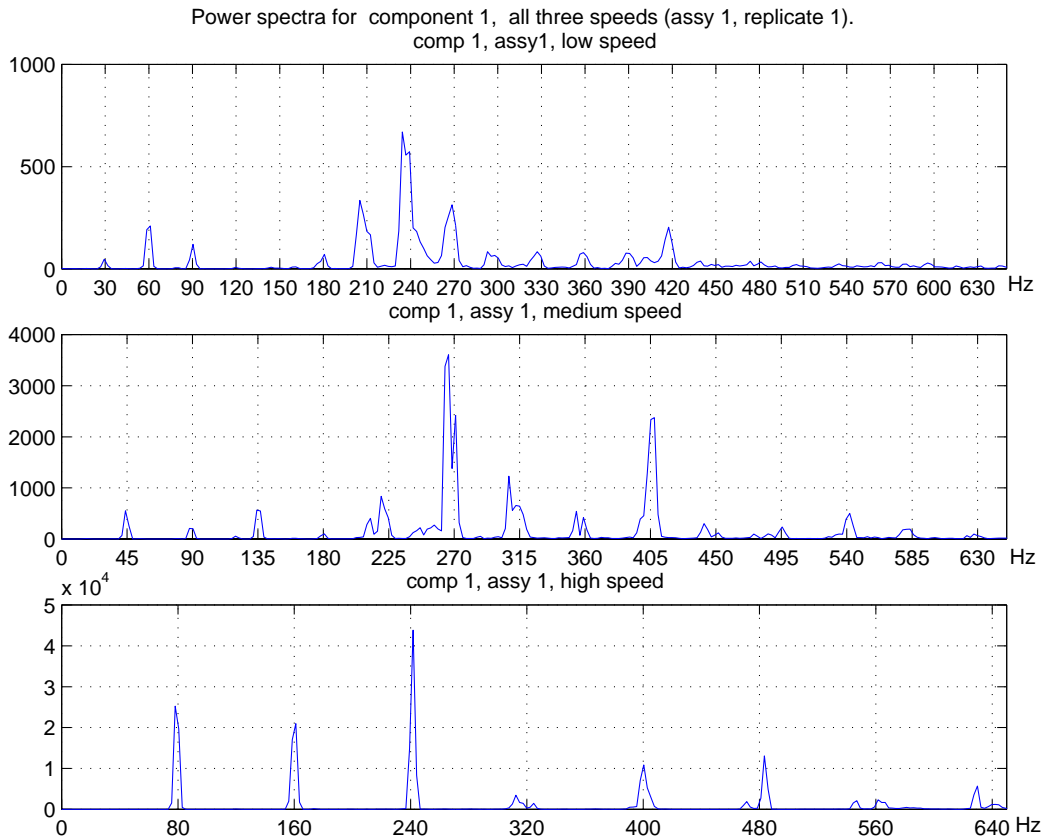


Figure 4. Power Spectra Comparison for Low, Medium, and High Speeds.

these shaft harmonics dominate the spectra, possibly masking any of the smaller effects that would be indicative of defects. However, a detailed analysis of the spectra for low and medium speeds shows a couple of frequencies that are not multiples of the shaft frequency, corresponding to the fault frequencies computed in Table 1. Specifically, the low speed spectra show a frequency component in the surroundings of  $90\text{Hz}$  and another frequency component around  $150\text{Hz}$ . Similarly, the medium speed spectra show additional frequency components in the surroundings of  $135\text{Hz}$  and  $225\text{Hz}$ . From Table 1, the first outer race fault frequency should be at  $89\text{Hz}$  for the low speed runs, and  $134\text{Hz}$  for the medium speed runs; the first inner race fault frequency should be at  $150\text{Hz}$  for the low speed runs, and  $225\text{Hz}$  for the medium speed runs. Slight deviations from these computed frequencies are typically expected due to variations in speed, geometry, and other conditions that cannot be accounted for. The magnitudes of the shaft speed and their harmonics are too high to show any defect frequencies for the high-speed cases. In general, the spectra are dominated by the shaft speed and their harmonics, making it difficult to isolate defect frequencies in a conclusive manner.

### Analysis of Trends in Total Power

A general observation from the analysis above is that the total power in the power spectra varies for each of the bearing components, for each speed case, in a manner that is suggestive of some defects in the components. To investigate this observation further, the total power (area under the power spectrum) is analyzed for each test condition. The change in the total power for each component and assembly instances for the low, medium, and high speed cases is shown in Figures 5, 6, and 7 respectively. The X-axis in these plots corresponds to each test run, for each component and assembly conditions, with 11 replicates for each run. For the case of the low speed runs, shown in Figure 5, component 1 shows the lowest total power levels, with a consistent increase for component 2, which validates earlier observations from the power spectral analysis. Component 2 has some type of a defect that results in higher total power, possibly due to manufacturing errors. The total power levels are initially lower for component 3 than it was for component 2, but increase slightly for the rest of the runs. In particular, component 3 is shown in Figure 5 as manifesting a large amount of variability in the case of the third assembly. It is possible that an assembly error was

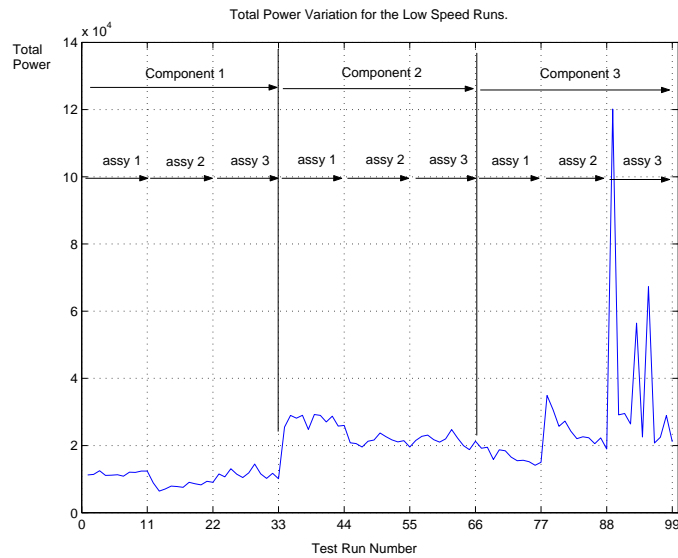


Figure 5. Total Power Variation for Low Speed Runs.

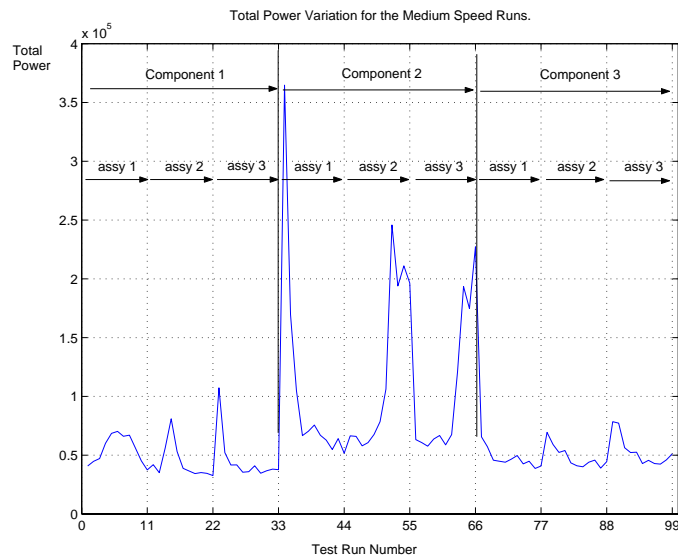


Figure 6. Total Power Variation for Medium Speed Runs.

introduced during the final disassembly and reassembly phase, hence resulting in the spectral energy to vary as the magnitude and direction of the impact forces on the bearings and shaft vary due to assembly errors. For the remaining components, it is difficult to observe any changes due to the different assembly conditions.

Figure 6 shows the same plot for the medium speed case. In general, more variability is introduced amongst the replicates as the shaft speed in the experiments is increased. The total power for component 2 is slightly higher than for components 1 and

3, just as was observed for the low speed runs. It is difficult to observe any differences due to the three assembly conditions. Finally, Figure 7 shows the same plot for the high-speed case. Note that the variability in the total power is so significant that it is difficult to isolate any specific patterns for the different bearing components. This excessive variability is due to the test-rig setup for this experiment, which excluded any loads on the shaft. The lack of a load was a requirement of the experimental design, to better isolate minute differences in the vibration characteristics of the bearings. Parallel work on helicopter transmission

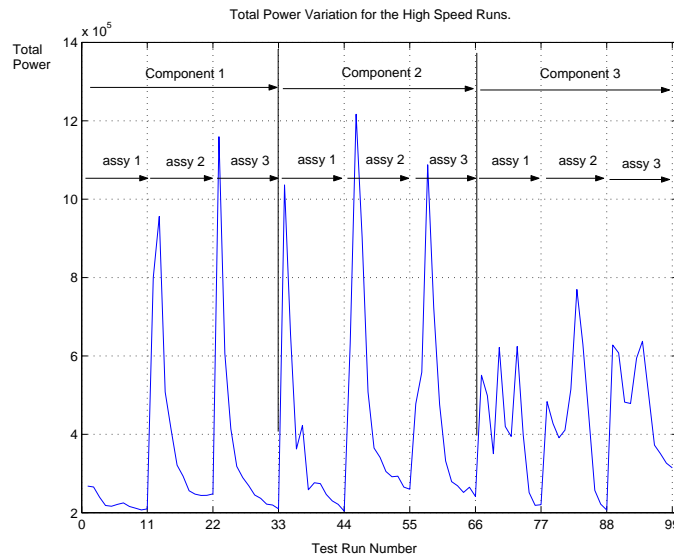


Figure 7. Total Power Variation for High Speed Runs.

vibration analysis also indicated variability for flight conditions where there was not enough load (ground and low-power conditions) (Huff et al., 2000a), reinforcing the need to conduct a second set of experimental runs with a load applied to the shaft. Because minute effects are still likely to be present in the current set of data, we next conduct a statistical analysis of the total power changes.

### Analysis of Variance in Vibration Characteristics

The analyses of the frequency content and total power for the test conditions show some preliminary trends that indicate that manufacturing defects (represented in the form of three different bearing components) have an effect on the vibrational energy. Since speed is a major factor in the amount of vibrational energy emanating from the bearing fault simulator setup, it is reasonable to expect a second and third order interaction between speed and manufacturing errors, and possibly assembly errors. An analysis of variance approach is used to study these interaction effects, as well as the individual effects and their significance for the vibrational energy. The response metric is total power of the power spectra (which equals the variance of the vibration series in the time domain.) The main assumption for an ANOVA approach is that the data are sampled from a normal distribution (Montgomery, 1991). Figure 8 shows a sample normal probability plot for the vibration data, indicating that the normality assumption is met for the originating data. In addition, the response metric used must also show normal characteristics. However, as observed from the analysis presented above, the total power varies greatly as a function of speed, hence casting some doubt to the validity of the results. A typical solution to such a problem is

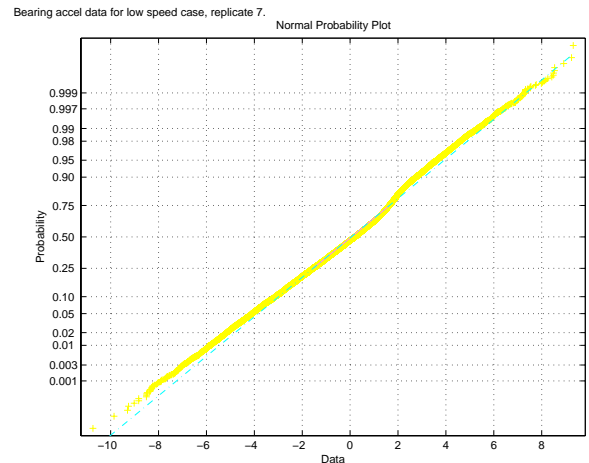


Figure 8. Normality Plots for Accelerometer Data.

to transform the response metric to make it behave as a normal distribution.

These initial results validate the general observations from the power spectral analysis. Table 2 shows the ANOVA results for the log-transformed total power metric, providing more reliable conclusions about the significance of each individual and interaction effect. *SS* is the sum-of-squares value, *DF* is the degrees of freedom, *MS* is the mean square value, *F* is the statistic value to assess significance, to be coupled with the significance level which indicates the probability that the effect is significant (Montgomery, 1991).

The results from the transformed metric indicate that speed

Table 2. ANALYSIS OF VARIANCE RESULTS FOR LOG TRANSFORMED TOTAL POWER.

Source	SS	DF	MS	F	Sig.
Intercept (hyp)	37221.742	1	37221.742	4047.495	.000
Error	18.687	2.032	9.196		
Speed (hyp)	454.178	2	227.089	101.344	.001
Error	8.314	3.710	2.241		
Assy (hyp)	.807	2	.404	1.439	.510
Error	.277	.987	.281		
Mfg (hyp)	18.224	2	9.112	4.154	.117
Error	7.843	3.576	2.193		
Speed*Assy (hyp)	1.467	4	.367	.904	.505
Error	3.245	8	.406		
Speed*Mfg (hyp)	9.118	4	2.280	5.620	.019
Error	3.245	8	.406		
Assy*Mfg (hyp)	1.277	4	.319	.787	.565
Error	3.245	8	.406		
Speed*Assy*Mfg	3.245	8	.406	3.188	.002
Error	34.352	270	.127		

is the main effect with a probability of  $\alpha = 0.001$ , followed by the third order interaction between speed, manufacturing errors, and assembly errors (probability of  $\alpha = 0.002$ ). The second order interaction of speed and manufacturing is the third significant effect with a probability of  $\alpha = 0.019$ . Manufacturing errors could be assessed as significant, if the acceptable probability level were set at  $\alpha = 0.2$  (significance level of  $\alpha = 0.117$ ). Assembly errors do not show to be a significant factor by itself, or in its interactions with the other two factors.

## DISCUSSION OF RESULTS AND FUTURE WORK

The analysis of the vibration data discussed in the above sections point to an initial correlation between manufacturing defects and vibrational signature changes. Specifically, a random sample of bearings drawn from a population of bearings is assumed to contain manufacturing defects in the form of surface waviness errors. The results above have shown a possible manufacturing defect in the second bearing used for the experiments. The analysis of variance based on the experiments indicates that manufacturing errors and their interactions with the different shaft rotational speeds can have a relatively significant effect on the total power in the vibrational energy. While pro-

viding us with an initial empirical correlation between manufacturing errors and vibrational energy, the difficulty in analyzing the vibrational signatures for specific defects is clearly demonstrated throughout this paper. This difficulty presents a serious roadblock to the development of tools to assess the performance characteristics of rotating components, at the development stage. The accepted solution is to provide separate inspection procedures before and after a product is placed in operation: manufacturing tolerances (set based on experience) are used to inspect products after development; vibration thresholds (set based on experience) are used to test the products prior to sending them to their final destination. There are two problems with this approach: 1) it is costly and time-consuming; 2) it is not reliable enough. The first problem is common to all industry, lengthening the production time and cost considerably. The second problem requires continuous monitoring and inspection of individual components throughout their lifetime, particularly when such components are produced for high-risk applications such as aircraft. In this work, we are trying to address both problems by establishing a correlation between the errors during development and the degradation in the performance of rotating components. This paper showed a study to determine whether standard methods can be used to establish such a correlation, using empirical

data, collected in a controlled test environment, using bearings as the main focus of the study.

Several important challenges were identified during this study. First, the high degree of variability in the experimental data indicates that the manufacturing effects are buried inside large factors which need to be taken into account. Currently, the methods and metrics used for such analyses cannot provide a reliable correlation between manufacturing errors and vibrational signatures, which makes it difficult to provide the correct information into the development cycle for early assessment of errors. Future work includes a measurement of the surface profiles of the bearings used in this study, test runs with pre-fabricated manufacturing errors, as well as analytical models of such correlations as a basis for comparison. Second, there is a lack of accurate analysis methods to enable the detection of features that are not masked by other factors and features in the vibration data. Future work includes a search for more reliable tools to detect minute statistical differences indicative of manufacturing errors. Third, there is a clear difficulty in quantifying and assessing the effect of assembly errors on the vibration signature. Future work includes developing a quantifiable metric necessary to feed such information into the product development cycle. Currently, assembly errors are being studied in a separate experiment, using an OH58 helicopter transmission test rig at the NASA Glenn Research Center, with the purpose of investigating reinstallation effects on the main pinion housing vibrations (Huff et al., 2000b). In the helicopter work, overall power levels show significant changes before and after reassembly, indicating the potential effect of assembly errors on the vibrational signature. Additional experiments are currently being planned using the MFS and the OH58 test rigs to address the rest of the challenges.

The work and results described in this paper demonstrate the necessity and need of addressing the problem of correlating manufacturing and assembly errors with vibrations during operation. Future work will concentrate on addressing the challenges identified above, so that automated tools can be developed to feed this information to the design and manufacturing engineers, in a systematic and reliable way.

## ACKNOWLEDGMENT

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